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TRANSLATION

INVESTIGATING PLASTIC
BEARINGS WITH INVERTED FRICTION COUPLING

By

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Investigating Plastic Bearings with Inverted Friction Coupling

by

A. A. Lebedev

The employment of plastics in the role of substitutes of non-ferrous metals and their alloys in friction units appears to be at present time one of the important problems, the solution of which offers greater technical-economical effect.

The essential shortcoming of plastics as bearing material is their poor heat conductivity. As result the friction heat is removed exclusively by lubrication and the body of the weight, which leads to overheating of bearing details, loss in strength of insert, disruption of the friction process.

The employment, in the role of lubricating and cooling liquid, of water does not always appear to be desirable and possible. Coating of metal inserts with a thin plastic layer is connected with a difficult technology of applying coatings and in a number of instances, e.g. in the presence of abrasive, it does not yield the desired effect.

In connection with this substantial interest is presented by a bearing with inverted friction coupling, in the antifriction material - plastics - is applied not on the insert, but on the shaft.

As shown by our investigations, such a bearing at proper structural execution, can work under much heavier friction conditions, than bearings with plastic inserts.

To manufacture experimental bearings was used wood pulp-lamellar plastic DSP-V with criss cross arrangement of the veneer sheet).Its physico-mechanical properties are as follows:

specific weight, g/cm^3...1.3 - 1.4; ultimate tensile strength, kg/cm^2...2650; ultimate compression strength, kg/cm^2...1750; elasticity modulus upon elongation, kg/cm^2...300000; impact strength, kg.cm/cm^2...80; relative elongation per unit length,%...1.0; hardness by Brinell, kg/cm^2...19 ; heat conductivity, cal/mhrs °C. ...0.4; heat resistance, °C....130-140.

Selection of DSP-V in role of antifriction material is explained by the fact, that this material has been most thoroughly investigated in friction units, which offers the possibility of comparing operating results with data of other researchers.

See page 2a for Fig. 1

F' ~. 1. Schematic of installation for testing bearings; 1-shaft; 2-insert; 3-dynamometer; 4-damper; 5-bushing; 6-spherical ball bearing; 7-holder; 8-cross piece; 9-support screw; 10-load.

A schematic of the installation, on which bearings have been tested, is shown in fig.1. The DSP insert was pressed on the shaft along the setting A_3/G and fixed additionally from the end with a clamping washer.

Mechanical characteristics of DSP drop considerably with rise in temperature (fig.2), consequently as maximum friction limits were accepted the ones, at which the temperature of the most heated zone of the bearing reached 70°.

The thermal effect of the bearing was controlled with the aid of eight copper-constantand thermocouples, piles of which were placed in a metal bushing 5 (fig.1) at a distance of 0.4 mm from the friction surface over the entire circumference. The identicity of thermocouple readings was checked by turning the bushing and by changing places of the thermocouples.

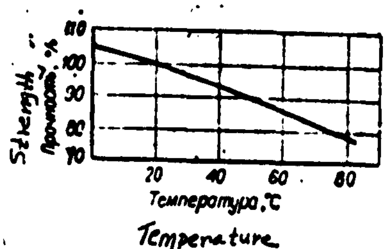


Fig.2. Curve showing change in strength of DSP in relation to temperature.

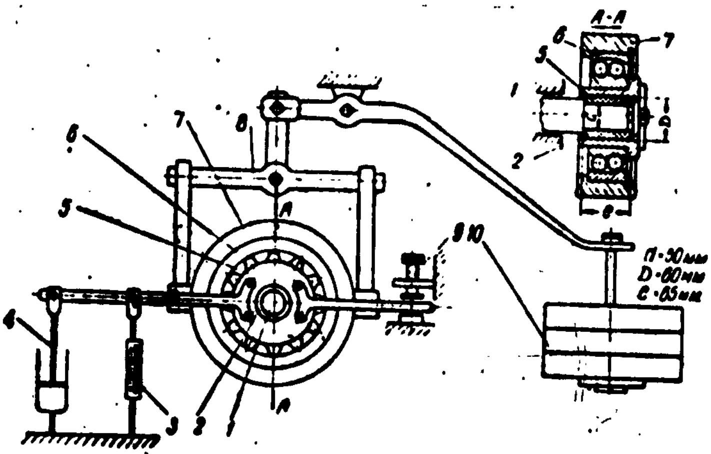


Fig. 1

In the role of lubrication were used oils: spindle oil 3 ($E_{50} = 2,9°$); machine oil C ($E_{50} = 5,6°$)- and water.

In fig.3 and 4 are shown graphs for the dependence of friction coefficients upon specific pressure and rate of sliding at various lubrication. As is evident from the graphs, in the greater range of loads and speeds the friction coefficient at machine oil lubrication is smaller than the friction coefficient with water lubrication,which is not being observed in ordinary bearings with DPS inserts.A comparison of data in fig.3 and 4 with experimental data, carried out by the scientific research institute of plywood, and the recommendations of the Orgchermet on the employment of DSP in the role of bearing material shows, that the employment of inverted couple noticeably increases the loading ability of the bearing.

See page 3a for Fig. 3

Fig.3.Curves showing change in friction coefficient in relation to specific pressure (v= 0.76 ./sec) at lubrication: 1-with spindle oil; 2-with machine oil; 3-with water.

As is known the load carrying capacity of a bearing depends to a greater extent upon the temperature of the most heated zone. In plastic bearings ,thanks to poor heat conductivity of the insert, the temperature along the perimeter changes particularly intensively, and in the zone of minimum gaps the insert is often overheated, which leads to loss in its strength, disruption of the oily layer,disruption of the friction condition and sometimes even to burning of the plastic.

In bearings with inverted coupling coming in contact with most heated zone plastics shrinkages are changing constantly, condition of cooling the plastic insert are

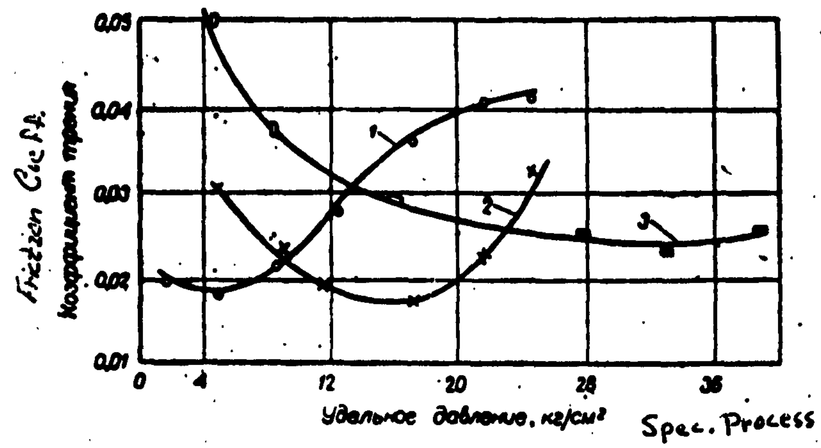

0,05
0,04
0,03
0,02
0,01
0
4
12
20
28
36
1
2
3
Коэффициент трения
Friction Coeff.
Удельное давление, кг/см²
Spec. Process

Fig. 3

improving, which assures more stable operation of the bearing under heavy friction conditions. Furthermore, the dissipation of heat in the zone of high temperatures follows at a more intensive pace.

See page 4a for Fig. 4

Fig.4. Curves showing change in friction coefficient in relation to rate of sliding (p=13.8 kg/cm²) at lubrication: 1-spindle oil; 2-machine oil; 3-water.

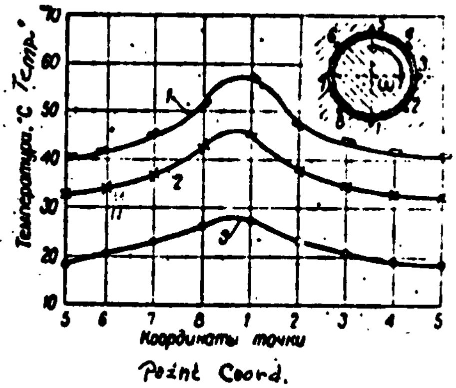


Fig.5. Curves showing temperature distribution along the perimeter of the bearing at specific pressures; 1-24.6 kg/cm²; 2-17.2 kg/cm²; 3-8.8 kg/cm².

The graphs in fig.5 characterize the temperature fields along the perimeter of the bearing with inverted couple at machine oil lubrication at specific pressures of 8.8; 17.2; 24.6 kg/cm² and circumferential speed v = 2.8 m/sec. As is evident from fig.5 the degree of irregular temperature distribution depends upon the specific pressure. But even at a specific pressure of 24.6 kg/cm² the temperature drop is 10-15° which points toward considerable stability of the thermal condition over the perimeter of the bearing. During water lubrication the temperature drop is 5-7°.

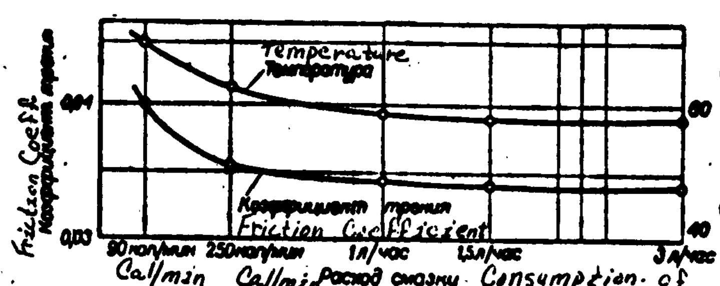
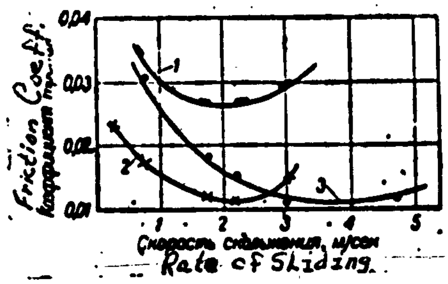


Fig.6. Dependence curves of temperature of most heated zone and friction coefficient upon lub consumption (spindle oil) at v = 0.76 m/sec and p = 13.8 kg/cm²

The investigated dependences of the friction coefficient and temperature of most heated zone upon the lub consumption showed, that for each specific pressure and velocity, i.e. for each condition, there is an optimum amount of fed lubrication of



Friction Coeff.
Коэффициент трения
0,04
0,03
0,02
0,01
1
2
3
4
5
Скорость скольжения, м/сек
Rate of Sliding

Fig. 4

given type. A further increase in the amount of lubrication does not give a temperature reduction of the most heated zone and in friction coefficient, in spite of the fact that the temperature of the outgoing oil decreases. And so, at $p = 13.8$ kg/cm^2 and $v = 0.76$ m/sec (fig.6) the rise in lub consumption to above 1.5 liters/hr shows no effect on the operation of the bearing. This, apparently, is true for all plastic bearings. But at an inverted friction coupling the amount of fed lubrication affects to a lesser degree the temperature of most heated zone, because as result of poor heat conduction of the plastic insert the heat transfer into oil decreases, and the role of lubrication as a cooling liquid decreases in this case.

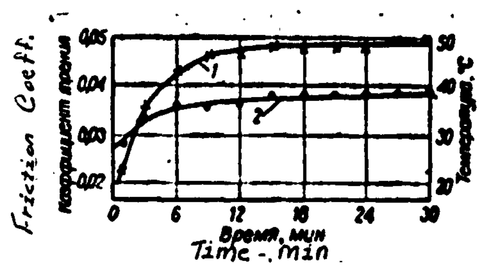


Fig.7. Temperature change curves (I) of most heated zone and friction coefficient (2) in the process of bearing operation ($v = 2.8$ m/sec, $p = 13.8$ kg/cm^2.

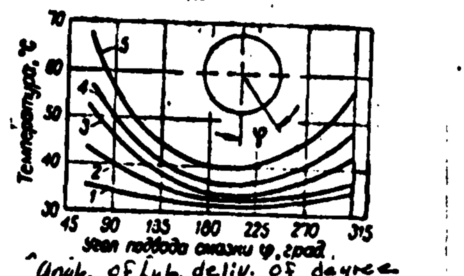


Fig.8. Temperature dependence curves of most heated zone upon angle of lub feeding φ and specific pressure p: 1-4.4 kg/cm^2; 2-6.9 kg/cm^2; 3-8.3 kg/cm^2; 4-11.2 kg/cm^2; 5-13.8 kg/cm^2.

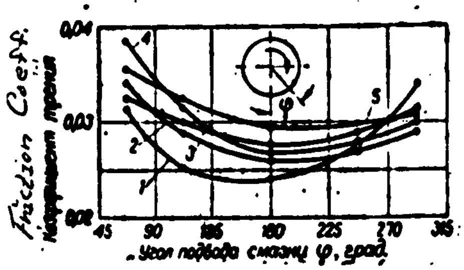
In fig.7 are given dependence graphs of friction coefficient and temperature upon operational time of the bearing beginning with the starting. As is evident from the graphs the friction and temperature coefficient are stabilized within 15-20 min, i.e. in bearings with inverted coupling the stable friction condition comes quite rapidly, which is due to more favorable conditions of heat discharge. The nature of dependence of friction coefficient and temperature is one and the same, which indicates a direct relationship between friction coefficient and temperature.

Investigated was also the ffect of the angle of delivery of lubrication on the temperature condition of the bearing and friction coefficient. Experiments showed (fig.8 and 9) that minimum values of temperature of most heated zone and friction coefficient correspond to the angle of lub delivery of 180-210°, whereby with the specific pressure increase of lub delivery effect, as evident from fig.8 and 9, it is possible to reduce the temperature of most heated zone by 25-30° and reduce the friction coefficient by 20-30%.

It was also established, that with reduction in relative gap (from 0.03 to 0.012) during oil lubrication the temperature of the bearing and friction coefficient decrease. At water lubrication the size of the gap has practically no effect on temperature and friction coefficient. After 80 hours of operation under nonstationary condition the plastic insert and metallic bushing showed practically no wear.

As result of conducted experiments it is possible to make the following conclusions:

1. The work of a DSP bearing with inverted friction coupling during lubrication with mineral lubrications of machine oil type is perfectly stable and gives a minimum friction coefficient at $v = 0.7 - 2.8$ m/sec and $p = 25-30$ kg/cm^2.



Angle of Lub. deliv. of degree

Fig.9. Dependence curves of friction coefficient upon angle of lub delivery φ and specific pressure (v=0.76 m/sec):
1- 4.4 kg/cm^2; 2-6.9 kg/cm^2; 3- 8.3 kg/cm^2
4-11.2 kg/cm^2; 5 - 13-8 kg/cm^2.

2. General method of investigating thermal condition by temperature of outgoin oil appears to be inacceptible to study of plastic bearings, because the determinant factor in this case appears to be the temperature of most heated zone over the perimeter of the insert.

3. For each friction condition there is an optimum amount of fed lubrication of given type. A further increase in the amount of delivered lubrication does not offer a temperature reduction of most heated zone in the bearing and reduction in friction coefficient.

4. The optimum angle of lub delivery from the view point of reducing the friction coefficient and temperature of most heated zone appears to be the angle of 180-210°.

5. With a reduction in relative gap from 0.03 to 0.012 at oil lubrication the temperature and friction coefficients decrease.

At water lubrication the effect of gap size on temperature and friction coefficient has not been revealed.

6. Investigation showed, that the use of inverted coupling in a plastic bearing noticeably improves the thermal condition of the bearing, increases its loadability.

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